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An Experimental Study of Heat Transfer and Pressure Drop in a Large Hydraulic Diameter Annulus

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ABSTRACT

An experimental investigation was carried out to determine the heat transfer and pressure drop characteristics in a large diameter annular test section for the hydro-dynamically developed and thermally developing laminar flow under constant heat flux condition. Studies were carried out under two conditions: with and without twisted tape inserts. The variations of friction factor and the local Nusselt number with Reynolds number for various twist ratios along the circumference of the test section were investigated. The friction factor has been found to be high for all the situations. The use of the twisted tapes has shown an improvement in the heat transfer coefficient. The present study will be useful in design of heat exchangers with large hydraulic diameters.

NOMENCLATURE

A	Area of cross-section, m^2
D	Diameter, m
f	Friction factor, non-dimensional
g	Acceleration due to gravity, m/s^2
Jg	Non-dimensional number
Nu	Nusselt number
Q	Discharge rate, m^3/s
q	Heat transfer rate, kW
Re	Reynolds number
T	Temperature, K
t	Time, sec
V	Average velocity of flow, m/s
X	Distance from leading edge of the test section, m
Y	Twist ratio (pitch/hydraulic diameter), non-dimensional
Greek Symbols	
μ	Coefficient of dynamic viscosity, kg/(m.s)
ρ	Density of fluid, kg/m^3
Subscript	
O	Outer
t	Total
Tt	Twisted tape
w	Wall
Superscript	
.	rate
"	rate

1. INTRODUCTION

Heat exchangers are used in various applications such as, desalination plants, marine appliances, solar collectors, etc. An estimation of the heat transfer rate in a laminar flow through a smooth annulus is often required in many appliances. Because of severe operating conditions as well as due to the flow of contaminated fluids, most heat exchanger surfaces are fouled with time. The formation of scale in long run results in a low heat transfer rate. One of the means to enhance the heat transfer rate is by inserting passive elements like twisted tape, wire coil, spring, ribbon, etc, in the flow field. However, increase in heat transfer rate increases the pressure drop in the duct as well, thereby requiring enhanced pumping power. Hence it is necessary to design such devices with a compromise between the enhanced heat transfer rate and pressure drop.

2. BACKGROUND

An extensive literature survey on heat transfer augmentation techniques with external inserts has been provided by Bergles [1]. Witham [2] suggested heat transfer enhancement by means of a twisted tape insert. Royds [3] observed that a twisted tape insert gives better heat transfer rate than a bare plain tube for low Prandtl number fluid. Cresswell [4] had observed that the ratio of maximum velocity to mean velocity is smaller in case of swirl flow compared to that for a straight flow. Date [5] made a review of the available friction factor and Nusselt number results for flow in a tube containing a twisted tape and pointed out that the existing correlations deviate from the measurements by 30%. The studies of Kepper [6], and Kidd and Jr. [7] had justified the usefulness of the twisted tape in a gas-cooled nuclear reactor. Date [8] formulated and solved the problem of fully developed, uniform property flow in a tube containing a twisted tape. Genis and Rautenbach [9] studied the thermo-hydraulic characteristics of high velocity water flow in short tubes with twisted tape inserts. Yokaya et al. [10] demonstrated a novel use of twisted tapes in controlling the flow in the continuous casting mold and refining process.

Klaczak [11] found the usefulness of the short-length twisted tapes. Hijikata et al. [12], while carrying out experiments in a vertical test section, observed that the radiation between the pipe wall and the twisted tape increases the heat transfer rate by about 50%. Saha and Dutta [13] found that short length twisted tapes are found to perform better than full-length twisted tapes for tighter twists. Further, Saha and Dutta [13] predicted that multiple twisted tapes provide the same heat transfer augmentation with higher-pressure drop as compared to a single twist.

In all the aforementioned studies the hydraulic diameter considered for study are small ($D_h < 30$ mm). To the best of the knowledge of the authors, no results have been reported for heat transfer and hydrodynamics pertaining to large hydraulic diameter pipe or annulus ($D_h > 30$ mm).

3. PRESENT INVESTIGATION

In the present work, experiments were conducted in the hydro-dynamically fully developed and thermally developing region of an annular test section with and without twisted tape inserts. The variation of the friction factor and the local Nusselt number with Reynolds number has been determined.

4. EXPERIMENTAL SETUP

Figure 1 presents a schematic view of the experimental setup. An overhead tank with 0.2 m^3 capacity serves as a constant head reservoir and is used to discharge the test liquid to the test section through a regulating valve. The test section consists of concentric straight pipes made of plexi-glass, which are joined at regular interval of 1.0 m by flanges. The inner diameter of the outer pipe is 50.0 mm and the annulus (flow passage) is of 30.0 mm in the radial direction throughout. The small pipe is supported inside the large pipe centrally by means of two small concentric cylinders made up of plexiglass of 0.3 mm thickness (Fig. 1). This concentric cylinder is supported circumferentially in the large pipe at each 2.0 m distance by means of three thin pins of diameter 2.0 mm each. The test section is kept horizontal by aligning it by means of a set of spirit levels.

The development length l_i for the laminar flow in an annulus is calculated using the following correlation Ozisik [14]:

$$l_i = 0.056\, D_h \,\mathrm{Re} \tag{1}$$

Where, $\mathrm{Re} = \dfrac{V\, D_h\, \rho}{\mu}$ is the Reynolds number, $D_h = D_2 - D_1$ is the hydraulic diameter of the annulus with D_2 and D_1 as the outer and inner radii of the test section. We have studied hydro-dynamically fully developed flow. The development length for the highest value of Re = 2000 considered and D_h = 30 mm is 3.36 m.

Pressure taps were placed at a distance of 4.0 m, 4.25 m and 4.75 m along the flow direction from the inlet valve position. At each location, three pressure taps were placed circumferentially at angles of 0^0, 120^0 and 240^0 measured from the top along the clockwise direction. For the study of friction factor, all the pressure readings are taken under isothermal and constant head conditions without any heat flux input to the test section.

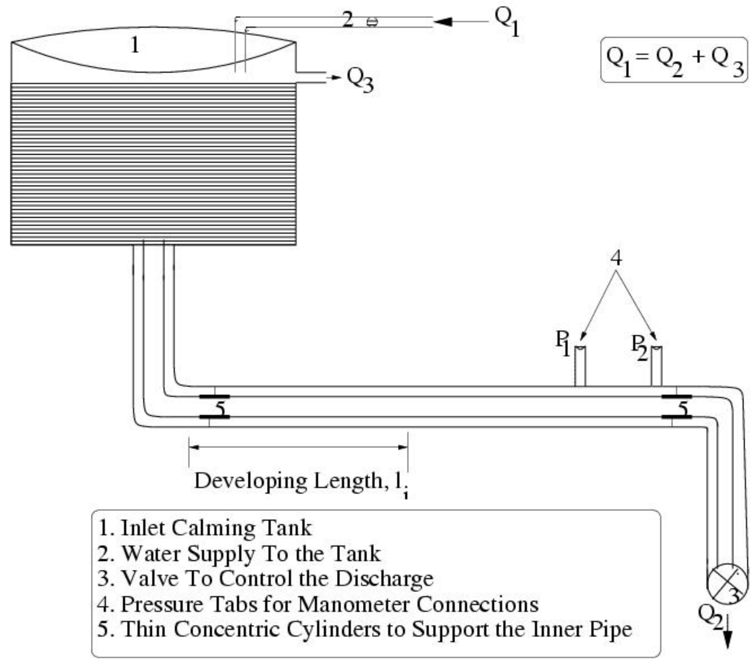

Fig. 1: A schematic view of the experimental setup.

A U-tube manometer with water as the working fluid is used for measuring the pressure drop at each probe location. The least count of the pressure measurement manometer is 0.5 mm. Water is used as the working fluid for the entire study. Water is supplied to the pipe from a big reservoir of capacity 700 liters, which is maintained at a constant head. The discharge through the pipe is measured by means of a calibrated jar and stopwatch.

The outer tube of the test section is made of plexi-glass and the inner tube is made of stainless steel. The length of the test section is 0.5 m. For measuring the temperature at different locations, thermocouples were used in the test section. A heater coil is put in the inner tube and the outer tube is insulated from the environment to reduce the heat loss. Three different heat fluxes were used in the present study, viz, 100, 150 and 200 KW/m^2.

The friction factor (non-dimensional) f in the developed region is calculated from the average pressure drop readings of the three circumferential pressure taps at three stream-wise locations (4.0 m, 4.25 m and 4.75 m from the inlet valve location) as given below (Ozisik, [14]),

$$f = \frac{2\,h_l\,g\,D_h\,A^2}{l\,Q^2} \qquad (2)$$

Where, h_l is the head loss noted from the manometer reading along the flow direction, A is the area of cross section and Q is the flow discharge rate through the test section.

The flow is fully developed hydro-dynamically but not fully developed thermally and the local Nusselt number is calculated as

$$Nu = \frac{h_{lo}\,D_h}{k_{lo}} \qquad (3)$$

Where, h_{lo} and k_{lo} are the local heat transfer coefficient and local thermal conductivity of the fluid, respectively, at a thermocouple location. The local thermal conductivity k_{lo} of the fluid is calculated from the fluid properties at the local fluid temperature. The local heat transfer h_{lo} is calculated from the energy balance at a thermocouple location:

$$h_{lo} = \frac{q''}{T_{lw} - T_{lmb}} \qquad (4)$$

Where, q" is heat input, measured by the wattmeter and is specified as per unit wetted area.

5. RESULTS AND DISCUSSION

The present experiments were conducted for constant water head and constant heat flux conditions. The parameters considered are given in Table-1.

Table 1: The values of the different parameters used in the experiment.

Sl No	Parameter	Values
1	Heat flux input	q = 2, 3.2, 4.2 kW/m^2
2	Twist ratio	Y=8.67, 9.23
3	Length of twisted tapes	L_{tt} = 0.3m, 0.62m
3.	Angular positions of measurement (along circumference)	0^0,120^0,240^0
4	Reynolds Number	40-2000
5	Twist ratio	Y=8.67 and 9.23

5.1 FRICTION FACTOR

First we report results on the friction factor under isothermal and hydro-dynamically fully developed condition Figure 2 presents the variation of friction factor with Reynolds number in large hydraulic diameter annulus and a plain pipe without the twisted tape insert. It is observed that the value of the friction factor f for the plain pipe is higher than that of the annulus. In the case of large hydraulic diameter ducts, the hydrostatic effect may be significant compared to the wall friction losses. Higher the amount of fluid in a given cross section, higher the loss due to velocity gradient arising due to the hydrostatic pressure. This results in a higher value of the total friction factor for plain pipe than a large diameter annulus.

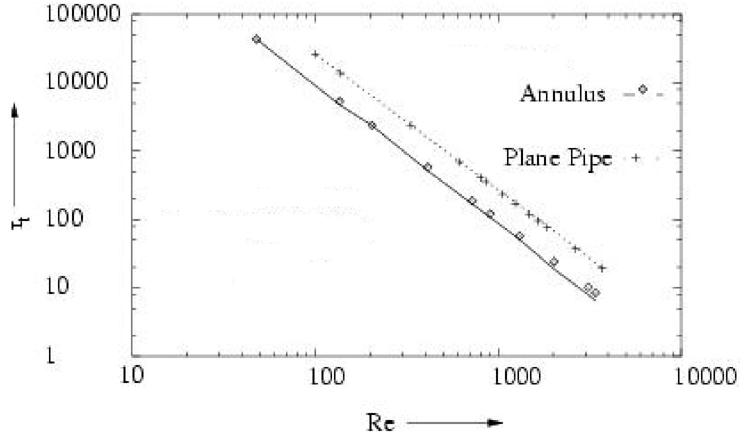

Fig. 2: Variation of friction factor f with Reynolds Number Re for plain pipe and annulus.

We propose that friction may be divided in two components, one due to wall friction and the other due to hydrostatic effects. The present experimental results shown in figure 2 yield the following correlation based on regression of data for the Reynolds number up to 2000.

For a plain pipe (without twisted tape):

$$f_t = \frac{26.79}{\mathrm{Re}} + \frac{529345.86}{Jg} \qquad (5)$$

For the annulus (without twisted tape):

$$f_t = \frac{75}{\mathrm{Re}} + \frac{6284.39}{Jg} \qquad (6)$$

Where, Jg is a proposed non-dimensional number and is the ratio of dynamic pressure to the shear stress produced by the velocity gradient due to the hydrostatic pressure variation along the radial direction in the test section ($Jg = \dfrac{\rho\,Q^2}{A^2\,\mu\sqrt{\dfrac{g}{D_0}}}$). The total friction factor decreases with an increase in Jg. With small hydraulic diameter and high Re, these correlations approach those given by Moody [15] and Olson and Wright [16]. However, for large hydraulic diameter and small Reynolds, the correlations (5) and (6) proposed by us seem to be more accurate compared to that proposed by Moody [15] and Olson and Wright [16].

A series of experiments were carried out to study the variation of the friction factor in a large hydraulic diameter annulus (D_h > 30 mm) with twisted tape inserts. Various insert parameters and different numbers of twisted tapes in an annulus for the fully developed laminar flow were considered. The laminar flow with varying Re from 40 to 2000 has been

considered. Twisted tapes of two different twist ratios Y = 8.67 and 9.23 and two different lengths l_{tt} = 62.0 cm and 30.0 cm were used in the study.

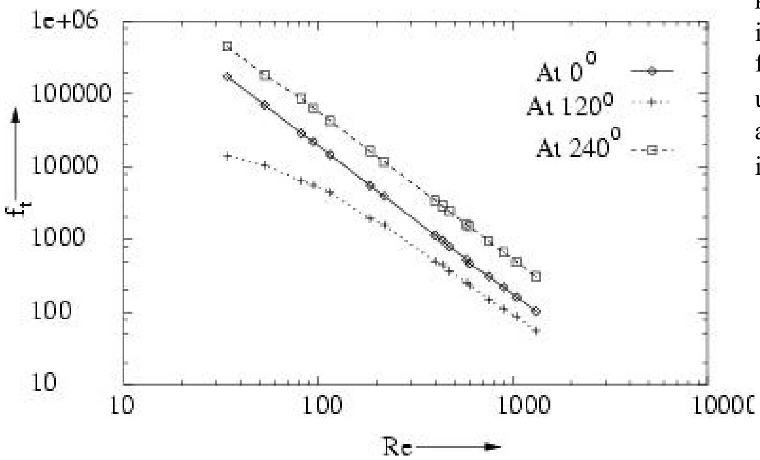

Fig. 3: Variation of friction factor f with Re for three twisted tape inserts at 0^0, 120^0 and 240^0 placed along the flow direction.

Figure 3 presents the variation of f with Re for three twisted tapes inserts (with Y = 9.23 and l_{tt} = 30cm) at 120^0, 240^0 and 0^0 along the test section. Here, f is found to vary almost linearly along the three angles considered. However the pressure drop is found to be highest at 240^0 along the test section, followed by that for 0^0 and 120^0.

It was found from the present experiments that the values of f for the three twisted tape inserts are quite high compared to the single twisted tape insert throughout the tested Re range. However the difference in f for two-twisted tape and three-twisted tape insert is found to be insignificant.

5.2 HEAT TRANSFER

Experiments were carried out under hydro-dynamically fully developed and thermally developing laminar flow condition with a constant heat input to determine the variation of the local Nusselt number Nu with Reynolds number Re in a large diameter annulus with and without twisted tape insert. Three different heat fluxes 2.0, 3.2 and 4.2 kW/m^2 were considered for the study. A twisted tape of dimension Y = 9.23 and l_{tt}= 30.0 cm was inserted at the leading edge of the test section along the 0^0. The variation of Nu with Re along the stream-wise direction was studied. All the experiments were carried out under the steady state condition.

Figures 4 presents the variations of local Nu with Re at the location $X = 0.02\ m$ of the test section along the three circumferential directions (0^0, 120^0 and 240^0). Results were generated with and without a twisted tape insert. Figure 4 is presented with an input heat flux of 2.0 kW/m^2. Experiments were also conducted for input heat flux of 3.2 kW/m^2 and 4.2 kW/m^2. It is found that the increase of Nu due to twisted tape insert is high along the twisted tape insert along 0^0. The reason for this is that the swirl flow generated by the twisted tape dies up quickly. The increase in Nu along other two angles (120^0 and 240^0) is found to be negligible. Further, with an increase in input heat flux Nu increases.

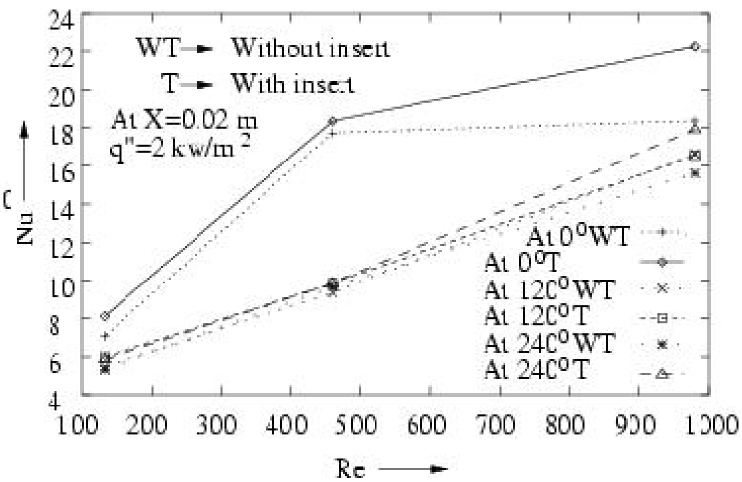

Fig. 4: Comparison of circumferential variation (at 0^0, 120^0 and 240^0) of Nu with Re with and without twisted tape insert (X=0.02 m, q''=2.0 kW/m^2).

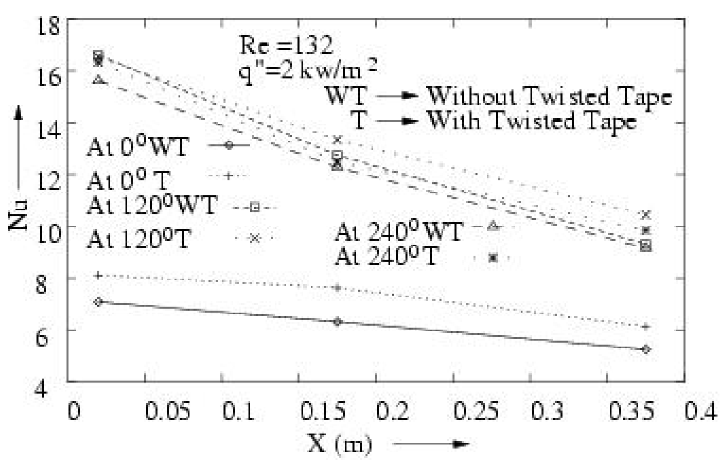

Fig. 5: Comparison of circumferential variation (0,120 and 240) of Nu as a function of distance from the leading edge of test section for uniform heat flux q''=2.0 kW/m^2 with and without twisted tape insert.

Figures 5 and 6 present the variation of Nu along the test section (along flow direction). These results are presented for circumferential positions 0^0, 120^0 and 240^0 for two

different heat fluxes 2.0 kW/m^2 and 3.2 kW/m^2, respectively. Further, results are given both with and without the twisted tape insert. The twisted tape insert was placed as tight fit along the flow direction.

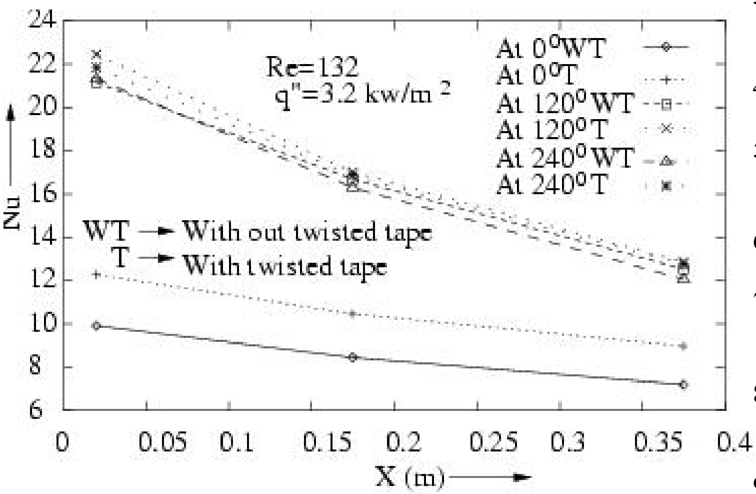

Fig. 6: Comparison of circumferential variation (0,120 and 240) of Nu as a function of distance from the leading edge of test section, for uniform heat flux q"=3.2 kW/m^2 with and without twisted tape insert.

It is observed that Nu decreases with distance from the leading edge as the flow is thermally developing. It is also observed that the increase in Nu due to twisted tape insert is large at the leading edge of the test section, but it goes on reducing in the downstream direction. This nature of Nu variation is due to decay of swirl flow along the downstream of the test section.

6. CONCLUSIONS

The following conclusions may be drawn from the present study:

1. The friction factor increases with number of inserts. However, the increment in the friction factor from two twisted tape inserts to three inserts is not significant.
2. The friction factor is found to be large along the twisted tape insert.
3. The Nusselt number in a large hydraulic diameter annulus is found to vary circumferentially for both with and without insert.
4. A significant increase in Nusselt number is noted along the twisted tape insert for large Reynolds number at the leading edge of the test section.

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